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INDUSTRIAL AIRCONDITIONING AND REFRIGERATION

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ENERGY EFFICIENT AIR CONDITIONING & REFRIGERATION

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The use of refrigeration and air conditioning is an essential part of electronic, food processing and hotel industry, where air conditioning is used to provide optimum working conditions for people, machinery or processes in order to improve the efficiency, safety, reliability or productivity. In some of the industries this load accounts for a very significant part of energy consumption.

PART - A

AIR CONDITIONING

1.0 GENERAL:

Air conditioning involves control of temperature, humidity, dust, gases, fumes, odours and bacteria in a working environment. The relative importance of these will vary from one application to another.e.g., Computers usually need a close control of humidity whereas food processing operations are more concerned with hygiene and cleanliness.

The air conditioning installation may therefore be required to supply air that is heated, cooled, dehumidified, humidified, filtered and scrubbed. There are two significant energy consuming stages involved to achieve proper air conditioning.

1. Refrigeration system to cool the air
2. Air handling unit to control the movement of air

2.0 BRIEF DESCRIPTION OF PROCESS

A typical air conditioning system with the refrigeration plant using an air cooled condenser is shown in Exhibit 1.

The air into the unit, normally a mixture of fresh air and recirculated air depending upon the internal and externalconditions,passes through filter unit, a cooling coil, heating coil if necessary, before it enters the conditioned space in the form of a supply air. The cooling coil, circulating the refrigerant to the compressor and from the condenser and expansionvalve, acts as an evaporator

3.0 FACTORS INFLUENCING ENERGYCONSUMPTION

3.1 The energy requirements of an air conditioning system are largely controlled by the quantity of air to be

conditioned. The energy needed for cooling Q is defined as :

$Q = (V \times C \times @ \times T) + (Mw + H_fg)$

Where Q -Energy supply or removal rate

V -Volume flow rate of air

@ -Air density

C -Specific heat capacity of air

T -Temperature difference of conditioned air

(in practice the air may have to be cooled and reheated if duhumidifying is required. These two temperature changes should be considered separately).

Mw -Condensation rate or evaporation rate of water in the conditioned air

Hfg - Latent heat of vapourisation of water

The power required by the chiller for air cooling will depend on the actual COP of the installation.

$W = Q/COP$

WhereW = Power requirement

Q = Energy removed by chiller

3.2 Energy used in circulating air around a ducting system is also dependent on flow rate of air. The energy required by circulating fans may be defined as ·

$$W = \frac{V X P}{\text{Equivalent efficiency}}$$

Where W= Power required

V = Air volume rate

P = Pressure rise required

4.0 PLANNED PROGRAMME FOR ENERGY SAVINGS

There are several areas where in practice immense energy saving potential exist.

In order to minimise the energy use of an air conditioning system, it is necessary to reduce the load on the system to minimum levels- The air flow rate, the temperature change in the air and the moisture added to or removed from the air. It is also necessary to ensure that the equipment is working as efficiently as possible to avoid excessive losses in meeting the actual demand on the system. The two different stages involved namely chiller plant and air circulation systems are discussed below:

4.1 CHILLER PLANT

a) Correct sizing of the chiller equipment for the required duty would result in optimum loading of motors and compressors, resulting in savings in electricity consumption.

b) Regular maintenance and cleaning - particularly the evaporator and condenser (heat exchangers) could lead to considerable savings. Dirt on the heat exchanger surfaces reduces heat exchange co-efficient and reduces the COP.

This is a very important aspect, but mostly neglected.

Practical study in a plant revealed that a thorough cleaning of an A/C plant gave the following improvements :-

Airflow rate increased by about	- 32% more
System COP increasing by about	- 52% more
Compressor running time reduced by	- 22% less
Energy cost reduced by	- 22% less

The greater the degree of fouling, the greater the savings obtained.

c) Some typical examples of loss of efficiency, due to various factors are (Exhibit 2)

Cause	Loss in Eff.
– fouled condenser tubes	- 20% loss
– Poor air distribution	- 20% loss
– Gas in liquid lines	- 10% loss
– Cheap expansion volve system	- 25% loss
– insufficient defrost	- 10% loss
– evaporators too small	- 20% loss
– evaporators fouled with oil	- 15% loss
– poorly designed suction line	- 15% loss
– compressor operating near surge	- 20% loss

d) If the condenser can be operated at a lower temperature ie., by using cooling water instead of ambient air as a final cooling medium, the power required by the compressor would be very much reduced due to increased COP of the system.

4.2 AIR CIRCULATION SYSTEM

The efficiency of fan systems can be improved in the following ways :-

a) maintain and clean fans, ducts and filters to minimise pressure rise required at the fan. The pressure drop along the air duct is proprotional to $1/dia^5$ so an obstruction which reduces the duct diameter by half, raises the pressure drop by a factor of 32.

b) Speed controllers fitted to the drives of the circulating fans can reduce the power required by the motor, compared to the alternative of constant speed fans and damper controls to limit the air flow (Exhibit -3).

% rated flow	%PowerConsumption		
	Outlet Damper	Variable inlet vane	Variable speed drive
0	—	—	—
20	70	58	8
40	90	70	10
60	105	75	30
80	115	90	65
100	125	115	125

It can be seen that by using a variable speed drive for fans, the energy consumption can be brought down by 70 percent ie.,there could be 70 percent savings in energy, when the speed of the fan is kept at a rate which would give 60% rated flow. In case variable speed motors are not readily available, management can take immediate steps by varying flow rates by inlet vanes.

4.3 IMPROVING AIR FLOW

Pressure drops around a ventilation system can be reduced by improved free flow of air within the areas served by A/c and in particular by ensuring that vents and exhausts are not obstructed.

In order to determine ways in which the load on the space conditioning system may be minimised, it is necessary to understand the factors affecting the system and the sources of load. The following factors may contribute -

a) **airflow rates**

It is necessary to determine the required air change rate realistically, for each area served by the system. This is very important, as this determines load on the system.

b) **close off unused areas**

It is necessary to close off unused areas/air not requiring A/c. Flexible partitions would be very useful to cordon off portions when not being used.

c) **sealing off gaps**

It is necessary to seal all windows and door frames, ensure that windows and doors fit properly, and fix automatic closing devices. All air ingress and escape represents an increase in the load of the fan.

d) **lowering the ceiling**

In large areas, consider lowering the ceiling to reduce the volume of air to be conditioned.

e) **sources of cooling or heating load**

— Fit and maintain internal or external shading devices to minimise solar heating.

— Switch off machinery and lights off when not in use.
— Improve insulation between conditoned spaces and outside.
— Improve roof and floor insulation.
— Insulate hot pipes which pass through conditioned spaces, and air ducts.
— re-use exhaust air as part or all of air input to the chiller, where this is compatible with cleanliness and ventilation needs. The exhaust air would require less processing than fresh air.
— Usefreeair cooling wherever possible.
— Use high efficiency lights.

Table quantifying cooling and heating loads is depicted in exhibit-4.

PART - B

INDUSTRIAL REFRIGERATION

1.0 INTRODUCTION:

Refrigeration systems are widely used in industrial cooling, food retail and air conditioning applications. The basic principles relating to the industrial and larger air conditioning applications, apply to all the refrigeration systems.

Many of the owners and operators all too often do not consider the cost and efficiency at which the cold is being produced. Experience shows that running cost savings of 25 % are easily achievable. Before going to the expense of buying and operating a refrigeration plant, one should consider if there is need for refrigeration at all. Cooling towers can cool water to 30 °C even on the summer days and usually much cooler in places like Bangalore.

2.0 REFRIGERATIONS PROCESS:

There are several methods of producing cold but the vapour compression process, is the most extensively used in this country to any significant degree. Hence this paper is focussed mainly on this system.

In the vapour compression system energy is absorbed by liquid refrigerant boiling in a heat exchanger known as evaporator. This energy comes from the substance which is to be cooled - water, airbrine,etc,. Thecompressor, which is driven by a prime mover (usually an electric motor) raises the pressure and consequently the temperature of the refrigerant. The compressed vapour is then cooled and condensed in a heat exchanger called condenser and gives up its latent heat, usually to air or water. The liquified refrigerant then passes from high pressure (through a throttling device) to low pressure and back to the evaporator. The cycle is then complete.

Since this vapour compression requires mechanical energy, the greater the difference between the condensing and evaporating temperatures (pressures) the greater is the power required by the compressor for the same amount of cooling. For the same compression ratio, a less efficient compressor uses more power and produce hotter discharge gas.

C O P OF THE SYSTEM:

The efficiency of the system is known as the Coefficient of Performance (C O P for short).

$$C\,O\,P = \frac{\text{Useful cooling effect (kW)}}{\text{Total power into the system (kW)}}$$

In real systems, the compressor motor represents only some of the power input, and running cost. Pumps, fans, crankcase heaters all consume power and in some cases also contribute to the heat load on cooling circuit.

Carnot C O P=$T_E/(T_C-T_E)$

Where, T_E is the evaporating temperature(deg K)

T_C is the condensing temperature (deg K)

Typically the values of C O P range from 2 to 5 at full load. (Practical refrigeration cycles typically have a COP of about 60 % of that given by the carnot efficiency)

It can be seen from the above formula that as the temperature lift (T_C - T_E)increases, the efficiency falls - more power is requied to do the same amount of cooling and, for the same temperature lift, systems which operate at higher evaporating temperatures (high T_E) are more efficient than those which operate at lower temperatures (low T_E).

3.0 ENERGY CONSUMPTION AREAS:

Electrical energy applied to the compressor causes system operation. Electriciy is also required for auxiliaries - circulating refrigerants, water and brine through various heat exchangers like - condensors, cooling towers, evaporators and chillers, and gererally these auxiliary loads account for as much as 25 % at full load and can be even 50 % at partial load.

The main equipment, the handling of which directly affects the C O P of the system is discussed below:

EVAPORATORS:

Three principal types of evaporators exist in indian industry.

- Direct expansion
- Flooded shell & tube
- Recirculation

The usage of each type depends on the type, size and capacity of the system application. The energy consumption is related to the temperature lift. The actual temperature lift in a real system will depend on the low and high temperatures heat sources and sinks - e.g., chilled water and cooled tower and the sizes of the heat exchangers.

Large heat exchangers will give closer approach temperature and therefore the temperature lift and energy consumption would be lower.

Energy consumption of refrigerant compressor is minimised by having highest possible evaporating temperature for a partucular system. This can be achieved by maximising the evaporator surface and maximising over all heat transfer coefficient. This heat transfer coefficient on the refrigerant side can be maximised by keeping the refrigerant clean from oil build up and maintain the correct refrigerant charge. Lower levels of refrigerant than required, increases the compressor power as the compressor struggles to meet the demand effecting increased compression ratio.

An evaporator efficiency can be drastically reduced by frost accumulation. Energy efficient defrosting would be required in direct expansion system Energy efficient defrosting depends on three factors:

— Determining when defrosting is required.

— Using an efficient method of supplying the heat to remove the ice.

— Terminating the defrost when the ice is melted and the water has drained off the heat exchange surface

COMPRESSORS.

The purpose of compressor is to draw refrigerant vapour from evaporator, therefore lowering the pressure and causing the liquid refrigerant to boil, extracting heat from the load at the desired temperature. Pressure of the vapour must then be raised by the compressor to a level where the vapour can be condensed by the available cooling medium.

Reciprocating compressors are the most commonly used type of refrigeration compressor and are available for wide range of duties. The other compressors that are generally available are rotary vane compressors, screw compressors and centrifugal compressors.

Cylinder unloading:

The capacity of reciprocating compressor can be changed by 'unloading' cylinders. This is done by holding the suction valves open so no compression is done in the cylinders. (This unloading system is often used on medium and large machines to enable them to be started off load.) Using this method, the compressor capacity can be unloaded in fixed steps - typically 100 %, 75 %, 50 %, and 25% of full load. This is a fairly efficient way of operating at reduced load; a compressor with half the cylinders unloaded will consume about 53% of the power of a fully loaded machine over the same pressure ratio.

A more superior method of part load operation is by using variable speed motor drives which allow fully variable capacity control which in the speed and pressure limits of the compressor. Typically, turndown to 40 % of full load is possible without cylinder unloading. The reduced rpm improves the compression efficiency.

Effect of valves:

The efficiency of the compressor is affected by the performance of the valves, the valve design being optimised for the refrigerant and pressure ratio used. A compresor should not therefore be operated over a different pressure level or with a different refrigerant without first ensuring that the valves are suitable. Operating a compressor with valves designed for low temperature operation at high temperatures could result in a 5 - 10 % reduction in cooling duty and a 20 % reduction in compressor efficiency.

Performance of compressors at part load:

When refrigeration systems are operated at less than full load, while the C O P is marginally increased the frictional losses in the compressor become a high proportion of total power and will tend to reduce the C O P. Part load C O P is only a balance between increased efficiency due to a smaller temperature lift and decreased efficiency due to due to increased losses. This has to be evaluated from system to system considering the size and type of installation.

Part load control by gas bypass and throttling the gas at the compressor inlet should be avoided as it is highly inefficient.

CONDENSERS:

Three types of condensers are usually found in industry, out of which water cooled shell and tube condenser is used widely.

* Water cooled shell and tube
* Air cooled
* Evaporative cooled

a) Water cooled Condensers:

These are used for all types of compressors with the refrigerant almost always on the shell side with water in the tubes. To give good tube side heat transfer the water velocity should be as high as possible, consistent with reasonable pumping power and freedom from erosion. A water temperture rise of 5°C and a temperature

approach of 5°C between water exit temperature and condensing temperature are good targets to aim for.

For the minimum condensing temperature, and therefore lowest compressor power, the water should be as cold as possible - always assuming that the expansion device will allow low condensing pressures to be used.

b) Air cooled condensers:

The refrigerant condenses inside the tubes while air passes over the tube surface. Because the air to tube heat transfer is poor, the tube surface is invariably extended using metal fins. But many times these fins can be found to be fouled. A temperature difference of 14°C between the condensing temperature and air inlet is normally economically achievable. A regular maintenance schedule should be drawn up to regularly clean the coils.

c) Evaporative Condensers:

In this the refrigerant vapour is condensed in tubes which are continuously wetted on the outside by a recirculating water system while air is simultaneously blown over them. These have a similar over all performance to cooling tower/shell and tube conderser combinations. A large condenser will give a lower condensing temperature and hence lower running costs, but at the expense of higher capital cost - the balancing point resulting in the lowest overall cost will be different for each system.

Loss of Condenser efficiency due to Non condensibles:

Air and other non-condensible gases can get into refrigeration system in many ways, which reduce condensor efficiency. The most common reason is insufficient evacuation of vessels prior to initial charging or after maintenance. There can also be inward leaks and over a period of time large quantity of air can get accumulated. Also, generally, there will be a very slow break down of refrigerant and even this results in build up of non-condensible gases.

The build up of air other non-condensible gases results in high condensing temperaturs and high apparant liquid sub-cooling in the condenser and their effect on operating efficiency could be serious.

If the condenser has 15 % air and 25 % ammonia, and the absolute pressure is 15 bar, the partial pressure of ammonia is 12.75 bar because of air, the discharge pressure is 2.25 bar greater. If the evaporating temperature is -10°C, the increase in energy consumption would be 12 %.

A test will confirm the presence of non-condensable gas in a shell and tube condenser. The refrigerant vapour supply and the liquid drain lines must be isolated with the compresser off and a refrigerant pressure gauge fitted to a suitable tapping. With the cooling water left on, the condenser temperature will rapidly fall to the water inlet temperature and the water inlet and exit temperatues will become the same.

If there are no non-condensable gases present, the gauge pressure will be the saturation temperature at the water temperature of, say 10°C in our example 5.1 bar gauge With the non-condensable gases present, however, the pressure will be about 7.1 bar and give a clear indication of their presence.

The same technique can be used with evaporative condensers, only here the air fans are left on and the water spray switched off. In a non-condensable free condenser the gauge pressure should be the saturation pressure at the air temperatue. With air cooled condensers the fans are left running and again the air inlet temperature is used.

Variation in wet and dry bulb temperatures:

When the climatic conditions provide us with large difference between dry and we bulb temperatures, it would

be of advantage to go in for water cooled condensers,because of large drop in the condensing temperatures. But when the difference in dry and wet bulb temperature is low and within 3 to 5°C, then all the type of condensers would perform equally.

Effect of Condenser type on over all system efficiency:

The auxiliary pumps and fans associated with the different types of condensers can amount to a considerable percentage of total system electrical power. In general, air cooled condensers require the least auxiliary power compared to evapo.ative and cooling tower based system. But in larger systems this auxiliary cost of the cooling tower based systems many times offsets with the improvement in the C O P compared to air cooled system. Therefore, the selection of condenser largely depends on the size of the system, location and other climatic considerations.

COOLING TOWERS:

Cooling towers are the most popular source of water for refrigeration system cooling. The cooling effect is achieved by evaporating some of that water into the air, the latent heat required comes from the remaining water which is consequently cooled.

The efficiency of a cooling tower depends on obtaining a thorough mixing of the air and water streams, and poor efficiencies are almost always caused by blockages reducing the air or water flows. Some of the common faults are:

* Blocked water spray nozzles
* Blocked or obstructed elimination plates
* Blocked or obstructed packing.

All these problems are normally due to mineral deposition or algae growth. Mineral deposition is usually controlled by water treatment and periodic or continuous blowdown which limits the concentration of minerals in the water. Algae growth is controlled by dosing the water with biocides.

Exhibit 5 shows a typical automatic control system for controlling the operation of cooling towers.

The temperature sensor detects the necessity for cooling and controls the by-pass valve and fan in sequence. If the cooling water is below the control temperature, for instance when the machine is off load or switched off, the bypass valve opens and the cooling tower fan is switched off.

4.0 HEAT RECOVERY:

By providing Suction line heat exchanger:

Once the refrigerant has left the evaporator it ceases to do any useful cooling. It is often the case that the suction line from the evaporator to the compressor picks up a considerable amount of heat either because it is well below ambient temperature or because, in remote evaporator applications, of its long length. This heat pick up can be greatly reduced by preheating the suction gas with warm liquid from the condenser before it passes through the expansion valve in a suction line heat exchanger.

For suction line heat exchanger circuits with R12 and R502, the total cooling capacity increases with increasing superheat out of the heat exchanger but for R22 and ammonia, increasing the superheat decreases the total capacity once the superheat is more than about 5°C.

5.0 REFRIGERANTS:

Four halocarbon refrigerants R11, R12, R22 and R502 apart from ammonia (R717) have widespread use in industrial refrigeration application in India. Some of the principal properties of these refrigerants are shown in exhibit-6.

This table also shows the theoretical maximum (or ideal)efficiency of a single stage vapour compression circuit operating over the range -15°C to + 30 °C using each of the refrigerants compared with the carnot efficiency of 5.73. Also shown is the relative size of the compressors required - this is determined by the volume of vapour at the compressor suction.

In practice the refrigerants which require smaller compressors are relatively more efficient than is shown in the table because the mechanical losses represent a smaller percentage of the total power.

6.0 INSTRUMENTATION:

The purpose of instrumentation on refrigeration system is to enable their performance to be assessed and to assist in the diagnosis of faults.

The common instruments that would require are mostly

- Pressure gauges
- Temperature gauges
- Motor power analysers
- Anemometer

Presure gauges are particularly useful for determining the flow of water (or brine etc.,) through pumps and heat exchangers. Pressure gauges, or at least gauge connections, should be fitted to the inlet and exit of each item.

Thermowells should be provided in all the heat exhangers to facilitate temperature measurement. Compressor loading can be checked by a power analyser.

Measurements:

Each plant has its own requirements, and the following list gives a general guide:

1) The actual compressor suction and discharge temperature should be measured for each compressor. For multistage compressors this should be done for each stage.
2) The refrigerant liquid line temperature from each condenser.
3) On all shell and tube heat exchangers whether evaporators or condensors, the temperature both in and out of the secondary fluid should be measured.
4) On air cooled consensers the air inlet and outlet temperatures should be measured.
5) On air cooling coils both the air inlet and outlet temperature should be measured.
6) Compressor and other associated motor loading should be checked for both part load and full load.

7.0 SCOPE FOR CONSERVATION:

a) Load reduction:

The reduction of the load on the refrigeration system is usually achieved by the following methods:

* Insulation of chilled water and refrigerant pipework.
* Insulation of chilled storage vessels and providing covers.
* Locating chilled water storage vèssels away from the chillers to reduce ambient temperatures.
* Maximisation of the size of chilled water storage vessels to take advantage of 'off peak' electricity tarriffs if applicable and low demand periods to minimise maximum demand.
* Automatic control of plant.
* Two stage cooling by utilising river water or cooling tower if available for first stage cooling then using chiller plant for final cooling.
* Using the maximum chilled water temperature possible to minimise the chiller requirement.

b) Electrical power reduction:

Savings in the use of electricity can be made in the following ways:

* Minimisation of the use of cooling tower fans and by using natural cooling whenever possible.
* Using variable speed pumps in conjunction with temperature detectors sensing the load on the system and adjusting the water flow as appropriate. Such systems can be used with 'power saving' controllers to further reduce electricity consumption.
* Using 'power saving' controllers on the compressors to reduce the power absorbed when 'off load'.
* Using sequence control on multiple chillers.
* Using cooling towers rather then chillers whenever possible or using cooling towers and chillers for 2 stage cooling of the chilled water.
* Regular defrosting of fan coil units in cold room application to reduce the 'insulating' effect of the ice built up on the heat transfer surfaces.
* Prevention of corrosion on heat transfer surfaces such as condensate by the correct choice of material for the environment.
* Regular cleaning of heat transfer surfaces to remove dust accumulation on air side and descaling/sludg removed on water side.
* Correctly sized duct work to minimise fan power.

8.0 MONITORING THE REFRIGERATION PLANT:

Once a refrigeration system has been purchased, installed and commissioned, the operating efficiency and overall running costs will be largely determined by the effectiveness of the day to day monitoring.

Even when an outside maintainence contractor is used, the onsite plant operators are the only people who are able to detect faults in their early stages when they can significantly increase the energy consumption without preventing the system meeting the required cooling duty.

The plant log sheets provide an effective way of gathering data on plant performance but they are only of any

real use if the data recorded is accurate, the information is intelligently analysed, and any problems identified are followed up.

A specimen log sheet for the simple water chiller is sheen exhibit-7.

9.0 CASE STUDY 1(Performance Monitoring-an example)

Using the performance data contained in exhibit 7, the system performance can be analysed item by item.

The Evaporator:

The water side pressure drops are close to the expected values so it can be assumed that the water flow is near to the design flow. The evaporator duty can then be calculated by:

$$\begin{aligned}\text{Duty} &= \text{water flow} \times \text{heat capacity} \times (T_4 - T_5)\\ &= 25\ \text{kg/s} \times 4.18\ \text{kJ/kg}^{o}\text{C} \times (7.4 - 4.9)\\ &= 261\ \text{kW}\end{aligned}$$

The heat transfer effectiveness of the evaporator is given by the formula:

$$\begin{aligned}\text{Effectiveness} &= (T_4 - T_5)/(T_4 - P_1)\\ &= (7.4 - 4.9)/(7.4 - 2.9)\\ &= 0.56\end{aligned}$$

This effectiveness can then be compared with that calculated from the system design or commissioning data. If the effectiveness from the measured data is significantly lower than that from the design data the cause should be investigated.

The Compressor:

The compressor manufacturers provide performance data for their compressors either in tabular form or as curves in the form shown in exhibit-8. Using this with the measured evaporating and condensing temperatures(P_1 and P_2) we get a duty of 245 kW for a power input of 45 kW for ful load operation, after correcting for actual superheat and sub-cooling.

For reciprocating compressors at part load (cylinder unloading) the full load data can, to a reasonable accuracy, be scaled in proportion to the cylinders in use.

In out case, the duty from the compressor curves agrees quite well with tht calculated from the water flow. In general, agreemeht to within 15% can be considered satisfactory.

The actual power taken by the compressor can be calculated from the current taken. Assuming the usual three phase low voltage supply:

$$\begin{aligned}\text{Power} &= 1.73 \times \text{Volts} \times \text{Amps} \times \text{P.F} \times \text{Efficiency}\\ &= 1.73 \times 415 \times 92.5 \times 0\ 87 \times 0.83/1000\\ &= 48\ \text{kW}\end{aligned}$$

The power factor of the motor can be measured with the help of a power analyser or obtained from the manufacturer or derived from the data on the motor name plate. The efficiency is the combined mechanical efficiency of the motor and of the drive system between the motor and compressor. If there is a direct drive this can be taken as 100 % and 95 % for V belt drives.

The Condenser:

The condenser can be analysed in a similar way to the evaporator. In this case the effectiveness is given by:

$$\begin{aligned}\text{Effectiveness} &= (T_4\text{-}T_6)/(P_2\text{-}T_6)\\ &= (27.4 - 25.0)/(30.0 - 25.0)\\ &= 0.48\end{aligned}$$

For air cooled and evaporative condensers it is usual to use manufacturrs published performance data. This gives the design heat rejection capacity for the measured condensing temperature and ambient air inlet temperature (dry bulb for air cooled condensers, wet bulb for evaporative condensers).

CASE STUDY 2 (Changing condenser from air cooled to water cooled)

The plant has an air cooled condenser. The climatic conditions of a place might-as well dictate the selection of a condenser and can have significant effect on the energy consumption. When the difference between the wet and dry bulb temperatures is around 3°C, then the air cooled condenser would be alright. But when the dry bulb and the wet bulb is apart by nearly 10°C cooling water condenser would be economical. However, the following case has several other aspects which depicted as follows:

DATA:

1. Plant capacity - 50 tons
2. Refrigerant used – R-22
3. Chilling load – Chilled water at 6°C
4. Type of condenser – Air cooled
5. Condenser outlet temp – 60°C.
6. Evaporator Pressure (Compressor suction) – 50 2 psig
7. Evaporating temperature – 26°C
8. Condensing pressure (Compressor discharge) – 269.7 psig
9. Condensing temperature – 50°C
10. Heat content at evaporating temperature:
    Liquid = 18.17 Btu/lb
    Vapour = 108.0 Btu/lb
11. Heat content at condensing temperature:
    Liquid = 48.6 Btu/lb
    Vapour = 113.57Btu/lb
12. Actual temperature outlet of condenser= 60°C which means a superheat of 10°C
13. Net refrigeration effect = 59.4 Btu/lb
14. Heat of compression = 5.57 Btu/lb

15. Circulation rates obtained per ton = 3.64lb/min (200/NRE)

16. Heat equivalent of work of compression per TR:
    (15) x (13) = 3.64 x 5.57
    = 20.27 Btu/min

17. Theoretical horse powler per TR = $\frac{20.27 \times 778}{33.000}$ = 0.48

18. Actual horsepower drawn per TR
    (System running at 50% loading) = 36kW/(25 x 0.75) = 1.92 HP/TR

19. Theoretical COP (13 / 14) = 10.66

20. The actual horse power drawn could be around(Assuming over all efficiency of the refrigerant circuit, including compressor and motor to be 60%) = 0.48/0.6 = 0.80hp/TR =1 hp/TR (Approx)

21. But the present HP drawn is too high, drawing about 0.92 hp/TR. The system requires immediate attention. The reasons for this are :

    a) Very high refrigerant inlet temperature to evaporator/chiller. Which is 10°C higher than the saturated liquid temperature. Which means that the sensible heat of the refrigerant will have to be removed before actual evaporation can take place. This reduces the net refrigerant effect and increases the circulation rates of the refrigerant. It is advisable to keep the superheat as low as possible (below 5°C.)

    b) The actual compressor outlet temperature was around 75°C. Whereas the saturated vapour temperature for the present operating pressure is 50°C, meaning that the vapours are also condsiderably superheated, to the extent of 25°C after the compressor. For gas to air heat transfer coefficients are rather small and condensation could be effected significantly, for the present condenser.

    The superheating could take place due to various reasons, such as deterioration of the glass and thereby production of non- condensables, pressence of air in the system, inadequate refrigerant, fouling of condensers, etc., All these could have a significant effect on the system.

Steps Required:

1. In the present case it is suggested to purge the system and fill it with entirely new gas.

2. Shift the air cooled condenser outlet outside, presently it is located inside the room, where the environment temperatures arehigh because of process heat and also deposits of various chemicals cause fouling; and thereby improved effectiveness.

   The savings that can be achieved is around Rs.2,00,000 per annum for each of the compressors.

3. If water cooled condensers are used, the temperature of the refrigerant that could be achieved in Bangalore is around 25°C less than the present designed inlet temperature of 55°C(max). This will mean that the refrigerant is subcooled. Also at the expansion pressure, amount of sensible heat that to be transferred by the refrigerant would be reduced by 50%. (often total sensible heat only) giving an improvement in the net refrigerant effect by 12%, for the same power invested, or lesser circulation by 12%, at the same evaporating pressure and temperature. The savings in this regard would approximately be Rs. 46,000 per annum per compressor.

FACTORS TO BE CONSIDERED:

1. The Net Refrigerating effect gets reduced as evaporating temperature falls down.
2. The circulation rate of refrigerant per ton of refrigeration increases as evaporating temperature falls down.
3. As the evaporating temperature falls down, the horsepower per ton increases and the superheating of the discharge gas increases.
4. As evaporating temperature falls, cycle efficiency decreases.
5. The heat rejection in the condenser increases as the evaporating temperature falls.
6. The volume of the gas to be handled by compressor increases as the evaporating temperature falls down.
7. As the evaporating temperature falls down the volumetric efficiency of the compressor also drops.

10.0 ENERGY CONSCIOUS HOUSE KEEPING MEASURES:

1) Shut off Air conditioning in unoccupied areas, infrequently used rooms and in areas with sufficient natural ventillation.
2) Vent moist air from driers and hot air from photocopying machines directly to outside.
3) Turn off unneeded exhaust fans and kitchen exhausts.
4) Isolate particular sections with higher requirements of conditioning (or use local units) within larger areas with no requirement of air conditioning.
5) Avoid artificial cooling where not needed:- Increase airflow or cool outside air (e.g., morning & winter) to handle internal loads without cooling; check if there is a saving by doing this because of additional energy for air transport.
6) Eliminate unnecessary exhaust hoods an roof ventillatiors or reduce their size.
7) Arrive at minimum operating time for air conditioning:
   a) Adjust working hours of personnel,
   b) avoid occupancy peaks,
   c) organise office cleaning during normal working hours (or with reduced ventillation),
   d) concentrate night work in one section of a building.
8) No artificial ventillation in winter, temperate season: Open windows, eliminate artificial ventillation where air quality, noise level, internal loads, and heating system permit.
9) Minimise operating time in areas with varying occupancy:Avoid unneeded ventillation in assembly rooms, lecture halls, kitchens, restaurants, toilets. Shut down ventillation if not needed.
10) Discontinue humidity control (except working hours in winter): discontinue humidification, dehumidification during nights, weekends, temperate season and summer (if possible)
11) Minimise operating time of cooling :
    a) by reducing cooling in late afternoon b) by flushing with cold outside air at night (e.g., increase air flow to avoid cooling when extra fan energy is less than for chiller).
12) Check possibility for intermittent use of AC system: In some office buildings intermittent use (e.g., 10 min

off, 20 min on) is practiced with success.

13) Check building for negative pressure to avoid infiltration of outside air and dust.

14) Maintain required temperatures:
   a) Encourage use of heavier clothing
   b) Readjust thermostats to necessary level
   c) Minimise drafts for comfort

15) Use sunshades optimally

16) Do not open Windows:
Opening windows in ventillated spaces will greatly increase energy use. Minimise infiltration by closing doors, unused chimneys & vents.

17) Shut off Lighting :
Shut off lighting if unneeded or in unoccupied areas to reduce heating load on Air Conditioning.

18) Shut off heat producing machines in unneeded places.

19) Leave thermostats at predetermined level where uninformed misadjustment would upset proper regulation of the system.

20) Bill departments, tenants according to energy used, to encourage energy saving measures.

21) Reduce refrigeration load:
   a) Reduce display lighting on refrigerated cases. Turn off at night.
   b) Set thermostats in refrigerated spaces to higher temperatures, if possible.
   c) Load freezers immediately after receipt of product, before unnecessary warming.
   d) Control moisture sources to reduce defrost cycling.
   e) Coordinate traffic into refrigerated areas to minimise door openings and infiltration.

22) Clean components: clean filters, heat exchangers, ducts.

23) Check leaks in valves, steam lines, air ducts, duct work joints, doors to fan rooms, check tight sealing of louvers.

24) Check efficiencies: check component and system efficiencies at full and low load, analyse energy consumption in detail, and compare with measured values. Identify excessive demand and components with low efficiency.

25) Check air inlets whether closed or misadjusted by user of a room. In many cases this was done because of improper functioning of the AC system. The correction made by the occupants may cause diffiulties in other areas.

26) Periodically check control and regulation, flow rates, recalibrate sensors.

27) Readjust the AC system at least twice a year for optimum summer and winter operation.

28) Check and analyse summer heat production: In many cases it is surprisingly high.

29) Reduce air change rates : by reducing external and internal loads, can the air change rates be reduced ? Reduce as far as possible without series complaints.

30) Adjust cooling tower operation at lower outside temperature - shut off some of the fans to reduce air flow rates

31) Increase chilled water temperature to highest possible level.

32) Check seals for oil leaks - oil contamination of the refrigerant can result in as much as a 25 % reduction in the C O P for a centrifugal chiller

33) Set pressure controlls on compressor to lowest possible setting that provides adequate refrigerant supply to expansion valve

34) Use automatic controls to operate exhaust fans as needed

35) Install time switches for shut down, start up, part load operation

36) Achieve energy conscious occupant behaviour

37) Install individual coolant meters and bill according to energy used to produce and incentive for saving

40) Insulate and seal duct work especially in non- conditioned spaces Use optimum insulation thickness

41) Change air path to reduce air flow rates connect in senes instead of parallel if possible

Use exhaust air of suitable areas (no odor, low temperature) as inlet air of less critical areas (corridors)

42) Install automatic blow down valves on cooling tower

43) Install automatic shutoff for fans in cooling towers for lower outside temperature or in nights.

44) Reduce cooling tower operation in winter
Avoid cooling tower operation during night

45) Assure purity of refrigerant to maintain high efficiency

46) Install insulating glass or double paned glass. Replace high conductive window panes

47) Repair building to reduce infiltration :
Seal penetration, joints, windows, door frames
Install revolving doors, automatic door closures

48) Improve controlls on refrigeration system.

49) Replace motors and other equipment with low efficiency (e.g., over sized motors, undersized air handling units)

50) Avoid dew points humidification - Replace dew point humidification with humidistat controlled system.

51) If obsolete equipment is replaced, reduce pressure drop of ducts (elbows, discontinuities, cross section, shorter air path, air handling units, filters)

Reduce fan speed at the same time

52) Install two speed motors, variable speed drives, or several units for optimum part load operation of pumps blowers, fans & compressors.

53) Use lake or river water (if available) for direct cooling instead of cooling tower

EXHIBIT -1

TYPICAL AIR CONDITIONING PLANT COMPONENTS

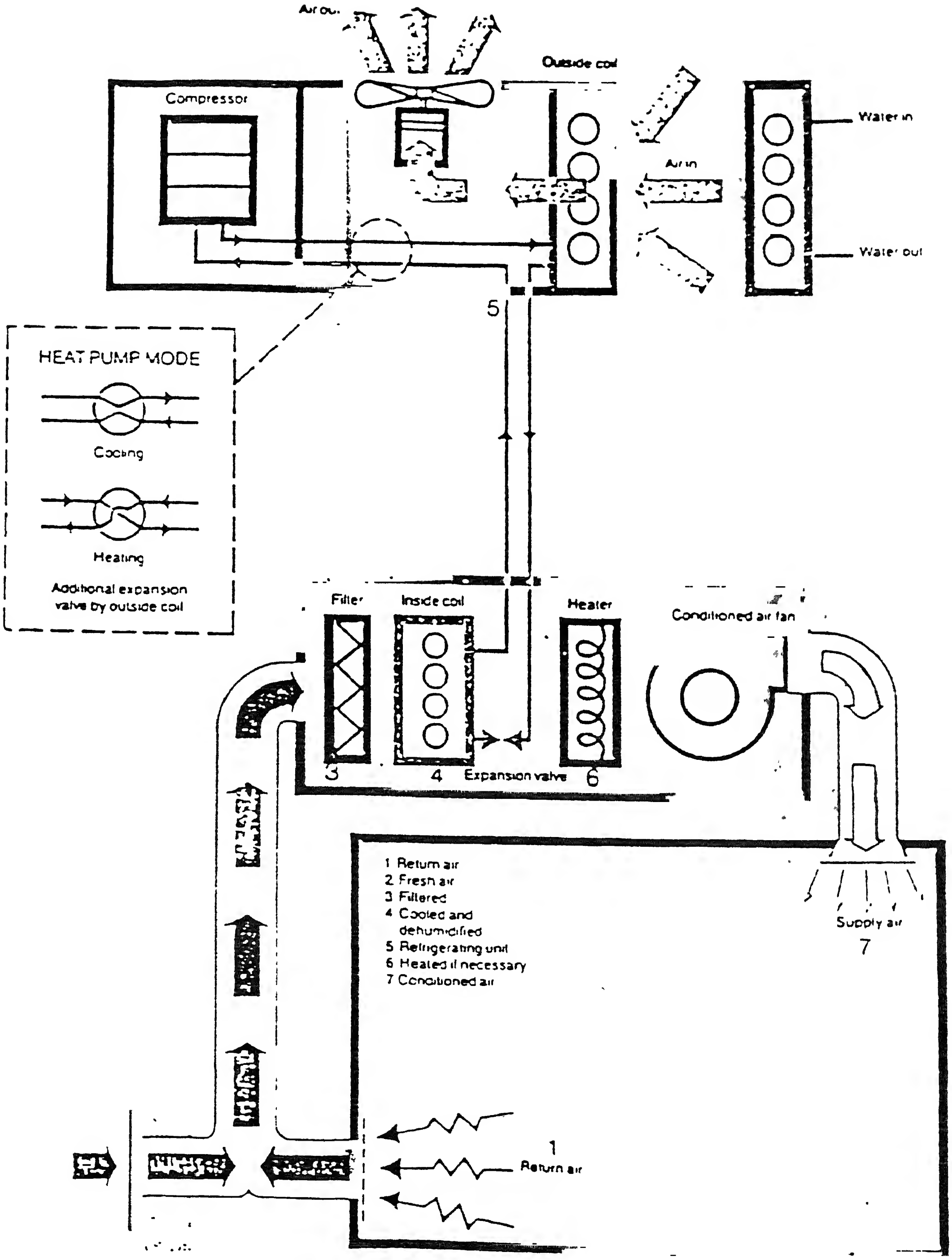
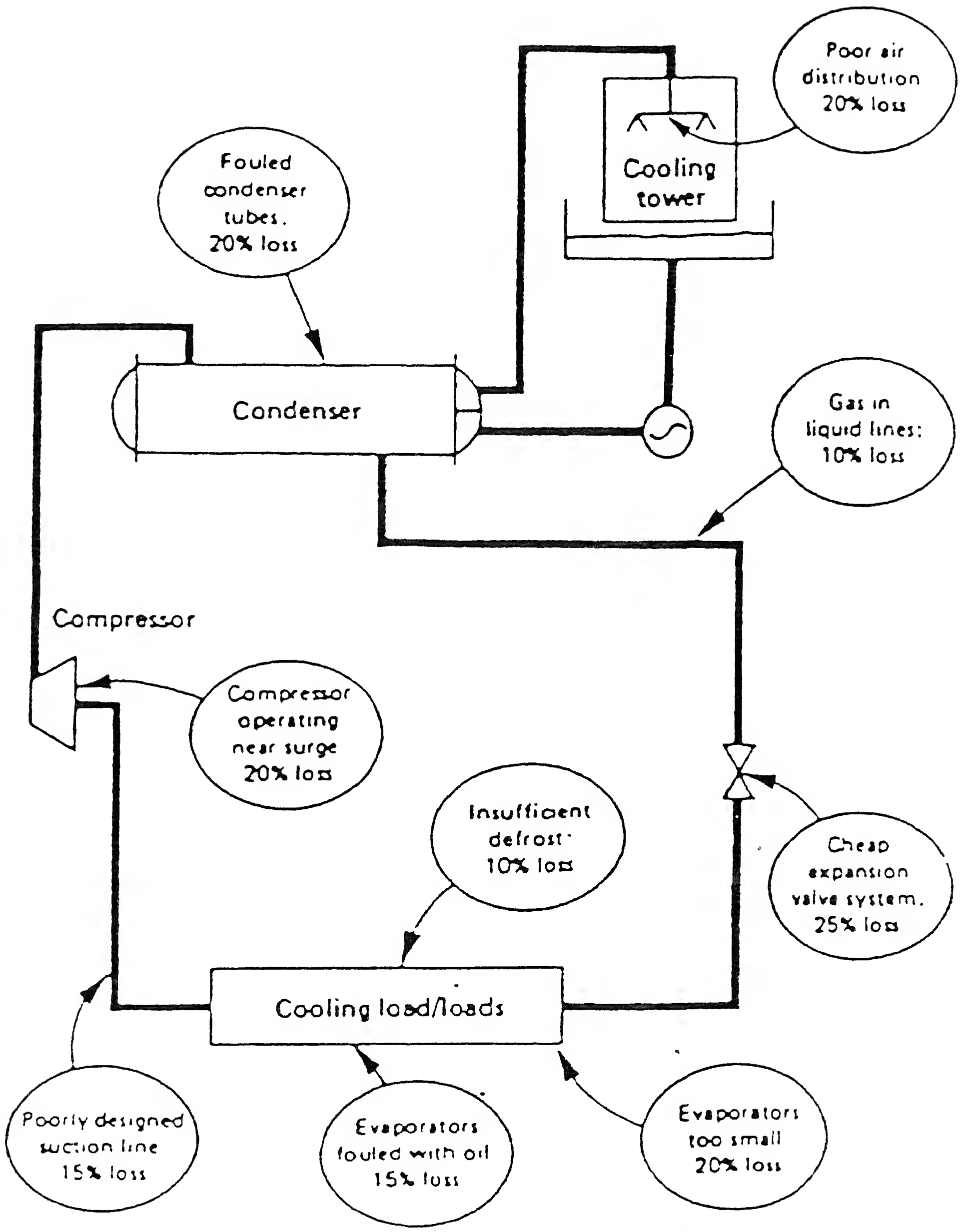


EXHIBIT -2



The diagram of a 'typical' refrigeration plant highlights those areas where losses often occur, and gives an average percentage loss of efficiency for each problem area.

EXHIBIT - 3

FAN POWER CONSUMPTION AT REDUCED AIR VOLUME

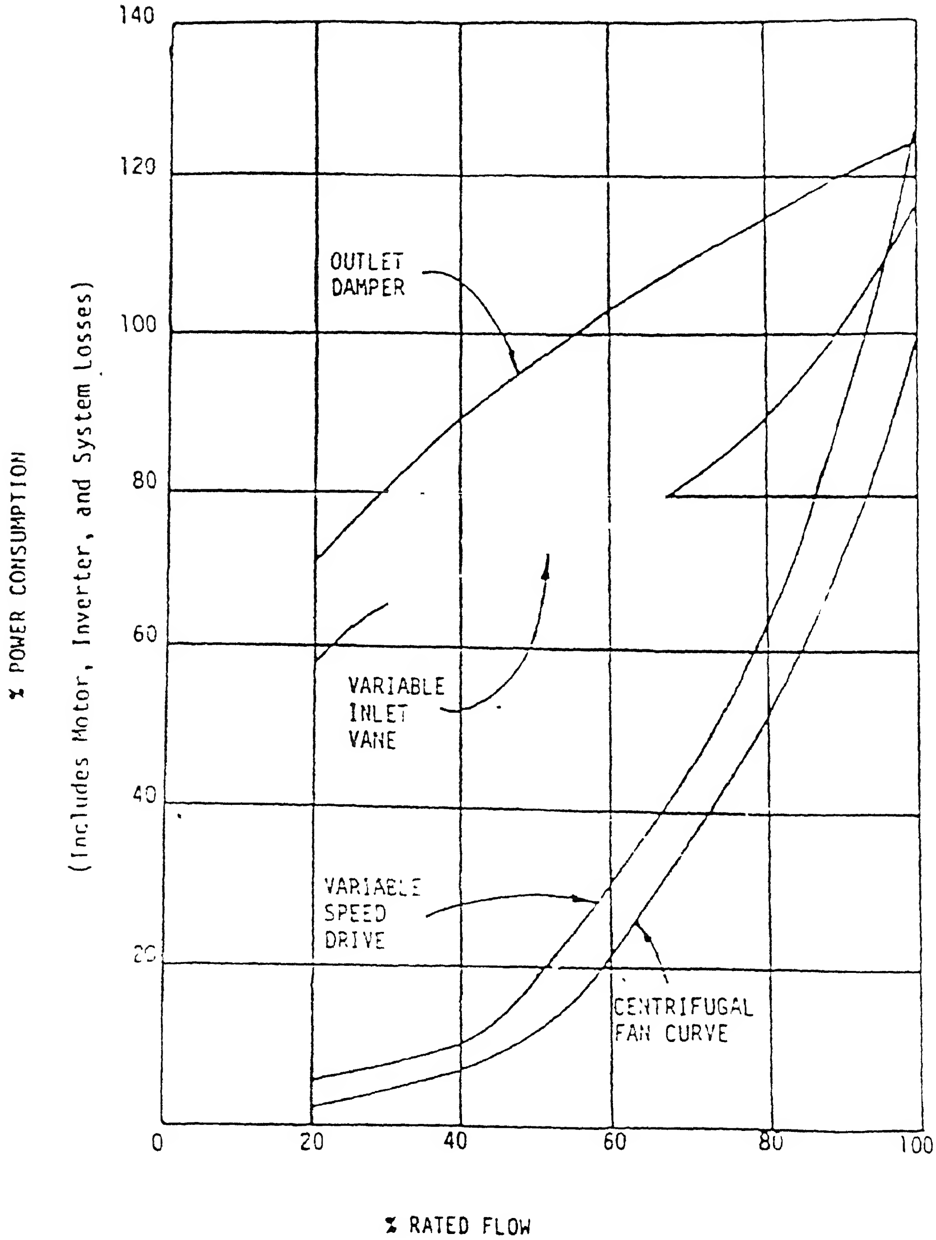


Exhibit-4

Heat dissipated per m² of floor area for rooms containing computer equipment

Room type	Sensible heat dissipated from equipment/(W/m²)	Typical equipment in room
Hi tech office	50-70	One VDU per person and one central printer
Computer support room	500-750	Distributive processors VDU Printer
Computer equipment room average high density	 450 900	Central processing unit Tape deck Disk drive

Heat dissipated from typical electronic data-processing equipment

Description	Performance/Status	Base dimensions/mm x mm	Heat emission/W
VDU terminal	–	470 × 590	200
	Intelligent	470 × 650	500
Printer	300 lpm	760 × 620	450
	430 lpm	930 × 810	1000
	750 lpm	930 × 810	1150
	1000 lpm	930 × 810	1150
Card reader	300 cpm	490 × 360	450
Disk drive	80 mb	480 × 860	980
	300 mb	580 × 910	1300
Magnetic tape deck	9 track	475 × 525	800
CPU	4 mips	640 × 830	3800
Photocopier	Running	Small	1500
	Standby	Small	750
	Running	Large	3500
	Standby	Large	1500
Typewriter			
electric	Running		50
electronic	Running		100

Notes: lpm = lines per minute, cpm = cards per minute, mb = megabytes mips = million instructions per second, cpu = central processing unit, vdu = visual display unit

Sensible heat and latent heat emissions from miscellaneous electric cooking appliances operatin under normal conditions of use

Appliance	Overall dimensions Less legs and handles (mm) (height quoted last)	Type of control	Miscellaneous data (Dimensions in mm, ratings in W)	Heat output (kW) Mfgr rating	Maintaining rate	In average use Sensible (s)	Latent (l)
Coffee brewer—2 5 litre		Man		0 7	0 1	0 3	0 1
,, warmer—2 5 litre		,,		0 1	0 1	0 1	0 1
,, brewing unit with 20 litre tank	500 × 760 × 660	Auto	Water heater—2000 W Brewers— 3000 W	5 0		1 5	0 4
Coffee urn—14 litre	380 diam × 870	Auto	Black finish	3 5	0 1	0 8	0
,, ,, 23 litre	460 diam × 940	,,	Nickel plated	5 0	1 0	1 0	0
,, ,, 14 litre	300 × 580 oval × 530	,,	,, ,,	4 5	0 1	0 7	0
Food warmer with plate warmer, per m² top surface		Auto	Insulated, separate heating unit for each pot Plate warmer in base	4 0	1 5	1 0	1
Food warmer without plate warmer, per m² top surface		Auto	Ditto, without plate warmer	3 0	1 2	0 6	1
Fry kettle— 5 kg fat	300 diam × 360	Auto		2 6	0 3	0 5	0
,, ,, 13 kg fat	400 × 460 × 300	,,	Frying area 300 × 350	7 0	0 6	1 1	1
Grill, meat	360 × 360 × 250	Auto	Cooking area 250 × 300	3 0	0 6	1 2	0
,, sandwich	330 × 360 × 250	,,	Grill area 300 × 300	1 7	0 6	0 8	0
Toaster, continuous	380 × 380 × 700	Auto	Two slices wide	2 2	1 5	1 5	[illegible]
,, ,,	500 × 380 × 700	,,	Four slices wide	3 0	1 8	1 8	[illegible]
,, pop-up	150 × 280 × 230	,,	Two slices	1 2	0 3	0 7	[illegible]
Waffle iron	300 × 330 × 250	Auto	One waffle 20 mm diam	0 8	0 2	0 4	[illegible]

EXHIBIT 5

EXAMPLE OF DIAGRAM FOR CONTROLLING COOLING TOWER FAN AND BY-PASS VALVE TO CONTROL CONDENSING TEMPERATURE

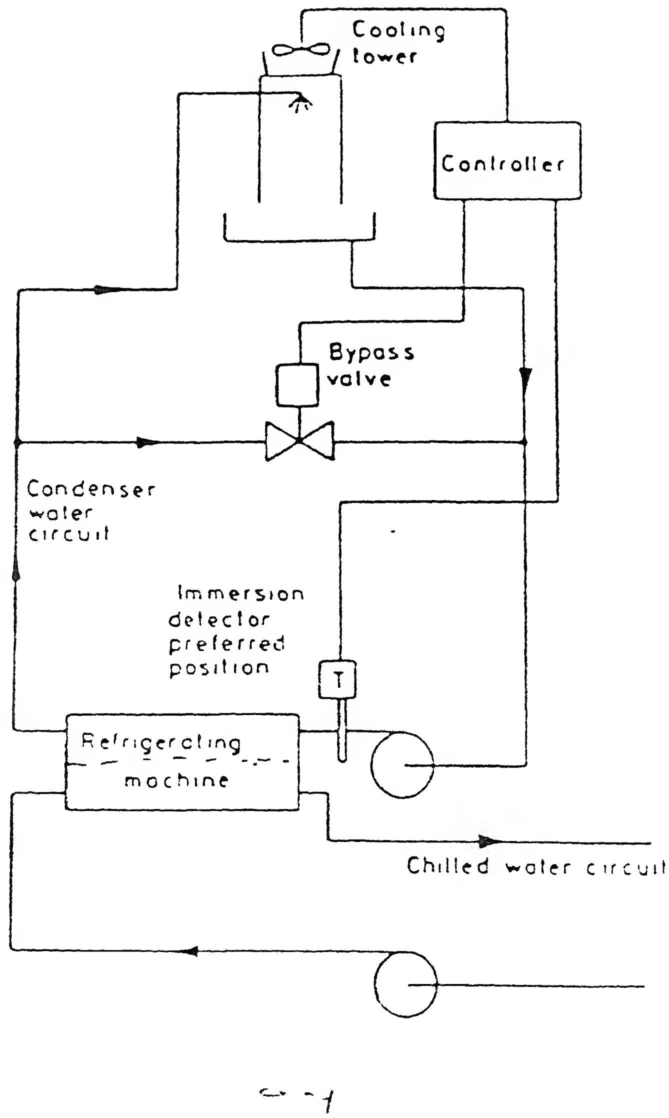
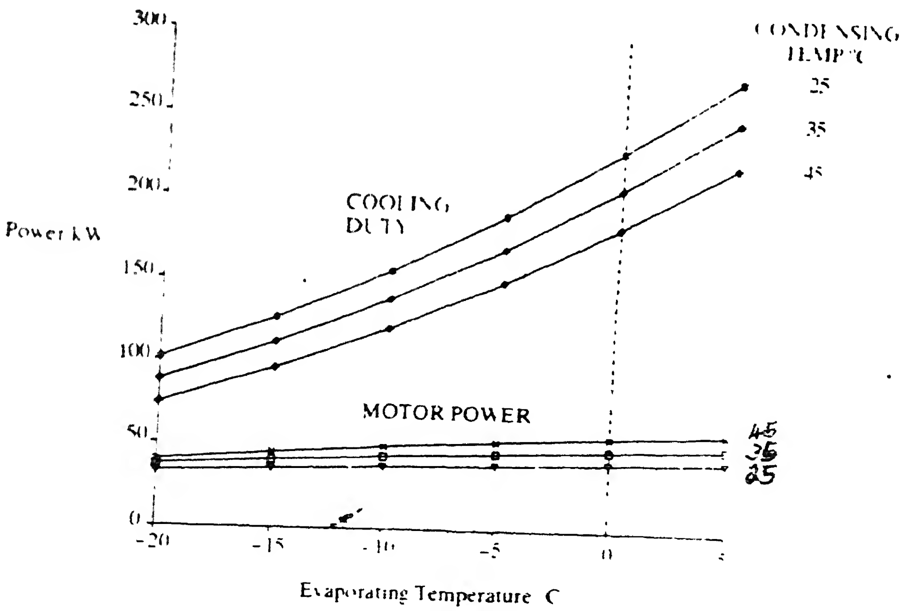


Exhibit - 6

COMPARATIVE PROPERTIES OF COMMON REFRIGERANT

Refrigerant	Saturation Pressure bar			Latent Heat @ 0 C	Specific Volume of Vapour @ 0 C	Ideal- 15% +30 C Efficiency Relative to Carnot Efficiency	Single Stage Cycle Relative Compressor Size for Fixed Duty
	- 15 C	0 C	30 C	KJ/Kg	m/Kg	%	
11	0.20	0.40	1.26	190.6	0.403	87.7	10.6
12	1.83	3.09	7.45	152.4	0.056	82.0	1.69
22	2.96	4.98	11.92	204.5	0.047	81.4	1.03
502	3.49	5.73	13.19	146.6	0.031	76.1	1.03
717 (Ammonia)	2.35	4.28	11.64	1262	0.290	78.5	1.0

Exhibit-8



TYPICAL COMPRESSOR PERFORMANCE DATA

Exhibit - 7

DATE	TIME	COMPRESSOR									
		HOURS RUN	AMPS	SUCTION TEMPERATURE SATURATED P1	SUCTION TEMPERATURE ACTUAL T1	DISCHARGE SATURATED P2	DISCHARGE TEMPERATURE ACTUAL T2	LIQUID LINE TEMPERATURE T3	COMPRESSOR LOADING %	OIL DIFF PRESSURE
RECOMMENDED VALUE ->				1° to 5°	P1+3° to P1+7°	T6+5° to T6+15°		P2-2° to P2-5°		> 3 bar
1/8 7 86	14 00	2332	[illegible]	3.0	6.1	30.4	57.2	28.0	100	4.2

SPECIMEN LOG SHEET

DATE	TIME	CONDENSER							EVAPORATOR					
		CONDENSING SATURATE P2	LIQUID LINE TEMP T3	WATER INLET PRES. P7	WATER INLET TEMP T6	WATER EXIT PRES P6	WATER EXIT TEMP T7	PUMP INLET PRES. P8	EVAPORATING SATURATED P1	WATER INLET PRES P4	WATER INLET TEMP T4	WATER EXIT PRES P5	WATER EXIT TEMP T5	PUMP INLET PRES P3
REC VALUE ->			P2-2° to P2-5°	2.8 bar		2.3 bar		0.8 bar	1° to 5°	2.5 bar	7° to 10°	2.0 bar	5°	0.8 bar
1/8 7 86	14 00	30.0	28.0	2.75	25.0	2.3	27.0	1.5	2.4	2.5	7.0	1.95	4.9	0.6

SPECIMEN LOG SHEET

SOME OF THE COMMON FAULTS ON REFRIGERATION SYSTEM

Major symptom	Other Symptoms	Fault	Solution	Operational Cost penalty
Low cooling duty compared with compressor curves	Bubbles in liquid line and low or zero sub cooling from	System under-charged LP float or TEV system	Add refrigerant to correct level	Upto 25% or more redn. in duty and COP
	On HP float systems	HP float valve stuck open bypassed. gas gas passing	Determine why bypass valve was opened initially. Correct fault and close by pass valve.	Upto 50% redn. in duty and COP.
	High actual compressor discharge temp & low compressor absorbed	Broken or obstructed reciprocating compressor	Repair valve and identify and rectify cause of	Loss of duty in proportion to cylinders affected.
Poor evaporator effectiveness	Low evaporating pressure high water/air side pressure drop	Fouling of air/water side of evaporator	Clean evaporator and locate and cure source of fouling	Upto 15% loss in COP 25% loss of cooling duty
	Low evaporating pressure high apparent super heat.	Blocked suction strainer	clean suction strainer. Identify and rectify source of blockage	Up to 15% loss in COP 25% loss of cooling duty
	Loss of oil from compressor crankcase	oil accumulation in flooded evaporator	Remove excess oil, install effective oil drain or recttification system	Upto 30% reduction in COP
	Loss of oil from compressor crankcase	Poor oil return from expansion valve system	Re-design suction side pipe work	Upto 25% reduction in COP

	All systems: possible high liquid line subcooling, high suction superheat	Obstruction in liquid line	Locate and clear obstruction. Identify cause and rectify	Upto 25% reduction in duty and COP
Poor condenser effectiveness	High condensing temperature, high liquid sub cooling	Very high over charge of LP float or TEV system	Remove excess charge	Upto 10% loss of duty, 15% reduction in COP
	High condensing high liquid subcooling	Air or non condensible gas in system	Purge non-condensible gas from condenser	Upto 10% loss in COP
	High water/air side pressure drop	Fouling of air/water side of condenser	Clean condenser and locate and cure source of fouling	Upto 25% loss in COP, 10% loss in duty
Low Suction Superheat	LP float and TEV: possible low compressor discharge temperature	Incorrect or faulty expansion device control	Identify and rectify fault	Upto 15% reduction in duty. Potential compressor failure due to liquid carry over.
High Suction Superheat	HP float: low liquid level in evaporator	System under charged	Add refrigerant to correct level	Upto 10% loss of duty, 71/2% reduction in COP
Low oil Differential pressure (Halocarbons)	Foaming of oil in crankcase particularly on start-up.	Refrigerant dissolved in crankcase oil in halocarbon systems due to crank case heater failure or liquid refrigerant in suction gas	Check operation of crankcase heater oil temperature should be 50-60deg. If heater OK check expansion device.	Potential compressor failure

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